HVAC Equations, Concepts, and Definitions

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A Few Acronyms and Definitions

Acronyms

- AFD – Adjustable Frequency Drive

- AFD Acronym Definition
- AFD A Few Days
- AFD Abbreviated Functional Description
- AFD Accelerated Freeze-Drying (food processing)
- AFD Accident Free Discount (insurance)
- AFD Acid Fractionator Distillate
- AFD Acoustic Flat Diaphragm (electronics)
- AFD Acrofacial Dysostosis
- AFD Acrofacial Dysostosis, Catania Type
- AFD Active Format Descriptor
- AFD Adaptive Flexible Defense
- AFD Adaptive Flight Display
- AFD Adjustable Frequency Drive
- AFD Advanced Full-screen Debugger
- AFD African Development Foundation
- AFD African Development Fund
- AFD Aft Flight Deck
- AFD Agence Française de Développement (French Development Agency)
- AFD Air Force Depot
- AFD Airfield Database
- AFD Airport Facilities Directory
- AFD Alarm Format Definition
- AFD Albany Fire Department
- AFD Alcohol Free Day
- AFD Alexandria Fire Department
- AFD All Friggin' Day
- AFD Alt.fan.dragons (Usenet newsgroup)
- AFD Alternative Forms of Delivery (Canada)
- AFD Amarillo Fire Department (Amarillo, TX)
- AFD American Funds Distributors, Inc.
- AFD Amsterdam Fire Department
- AFD Ancillary Function Driver
- AFD Angwin Fire Department (Angwin, CA)
- AFD Anticipatory Failure Determination
- AFD Apical Fibrobullous Disease
- AFD Approved for Design
- AFD Approximately Finite Dimensional
- AFD April Fool's Day
- AFD April Fools Day
- AFD Arc Fault Detection
- AFD Arc-Fault Detection
- AFD Architecture Flow Diagrams
- AFD Area Forecast Discussion (US National Weather Service)
- AFD Armed Forces Division
- AFD Arming & Fusing Device
- AFD Arming-Firing Device
- AfD Articles for Deletion (Wikipedia)
- AFD Ask for Details
- AFD assign fixed directory (US DoD)
- AFD Assistant Flight Director
- AFD Association Franèaise des Diabétiques
- AFD Athletic Field Design
- AFD Atlanta Fire Department
- AFD Austin Fire Department (Texas)
- AFD Autómata Finito Determinista
- AFD Automated File Designator
- AFD Automated Forging Design
- AFD Automatic Fault Detection
- AFD Automatic File Distribution
- AFD Automatic Fire Detection
- AFD Average Fade Duration
- AFD Away from Desk
- AFD Axial Flux Density
- AFD Axial Flux Difference
- AFD Active Format Description
- AFD Adaptive Forward Differencing
- AFD Adjustable Frequency Drives
- AFD Asus Foundation Drivers

AFA^{2}D

Acronyms

- AFD – Adjustable Frequency Drive
- AHU – Air Handling Unit
- ASHRAE – American Society of Heating Ventilating and Air Conditioning Engineers
- CV - Constant Volume
- HVAC – Heating Ventilating and Air Conditioning
- MOA – Minimum Outdoor Air
- Psych Chart – Psychrometric Chart
- VAV – Variable Air Volume
- VFD – Variable Frequency Drive
- VSD – Variable Speed Drive

AFA^2D

Definitions

- Sensible energy, Q_S (Btu's, Btu's/lb)

  Energy that causes a temperature change we can feel

- Dry bulb temperature, T_{db} (°F)

  An indication of sensible energy measured by a standard thermometer exposed to air; increasing dry bulb temperature = increasing sensible energy

AFA^{2}D

Definitions

- Latent energy, Q_L (Btu's, Btu's/lb)

  Energy that is used to keep water in a vapor state

- Wet bulb temperature, T_{wb} (°F)

  An indication of latent energy measured by a standard thermometer with its bulb covered by a wick that is saturated with water and exposed to moving air; increasing wet bulb temperature = increasing latent energy

AFA^{2}D

Definitions

- Dew point temperature, T_{dp} (°F)

  The temperature at which water will begin to condense out of a given sample of air. Also an indication of moisture content; increasing dew point = increasing latent energy.

  At saturation $T_{dp} = T_{wb} = T_{db}$

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Definitions

- Enthalpy, η (Btu/lb$_{dry\ air}$)

    A measure of the total energy content of air including both sensible and latent energy; increasing enthalpy = increasing energy content

AFA^{2}D

Definitions

- Relative humidity, RH (%)

  The amount of water vapor in the air at a given temperature relative to what it could hold at that temperature; 100% = saturation; increasing specific humidity = increasing moisture content, increasing dew point, and increasing wet bulb temperature.

  In Antarctica, the relative humidity approaches 100% much of the time, just like in Florida after a thunderstorm

AFA^{2}D

Definitions

- Specific humidity, ω (lb_{water}/lb_{dryair}, $grains_{water}/lb_{dryair}$)

  The ratio of the mass of water to the mass of dry air in a given sample of air; increasing specific humidity = increasing moisture content, increasing dew point, and increasing wet bulb temperature.

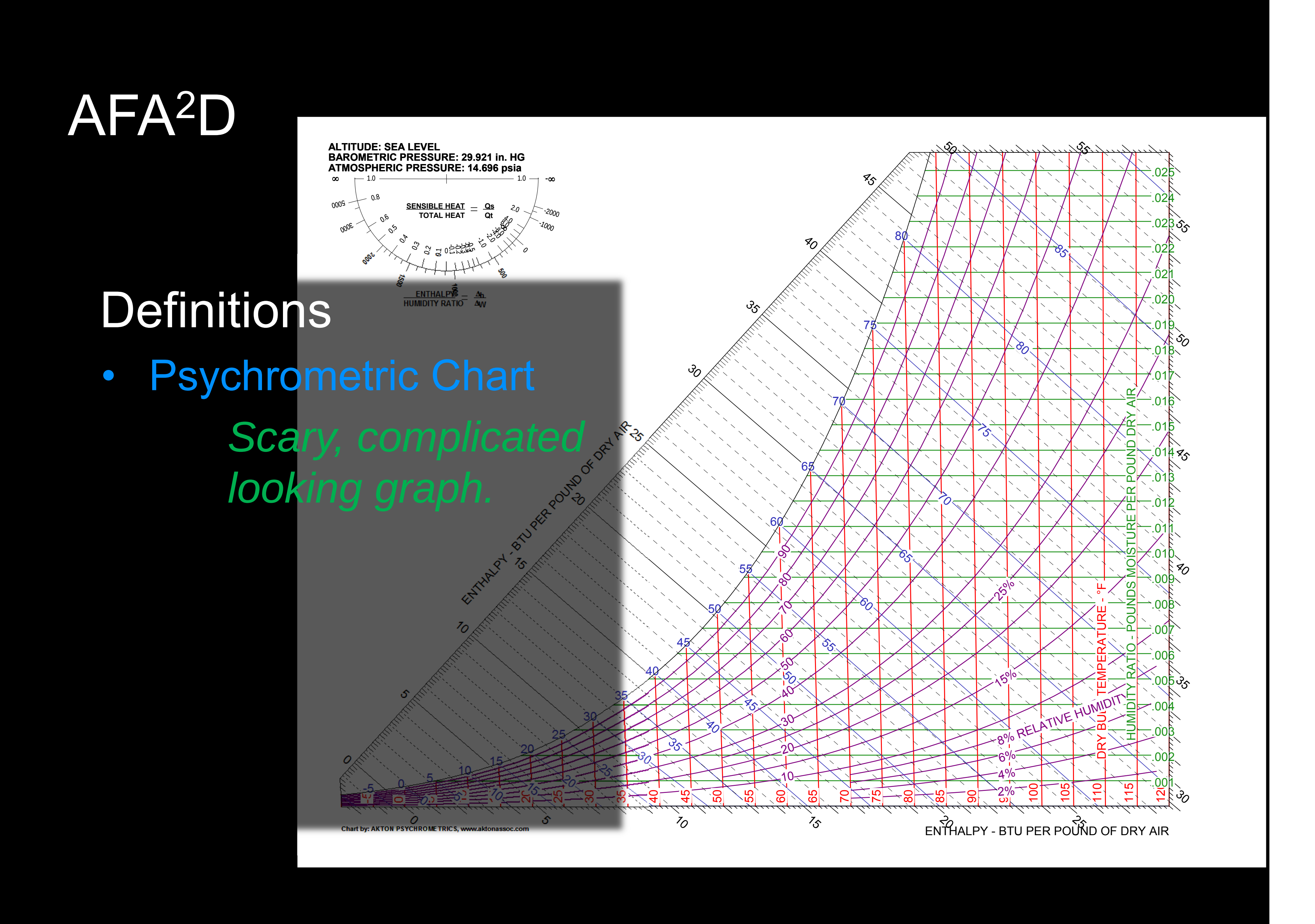
  In Antarctica, the specific humidity at a relative humidity of 100% is very low. In Florida, the specific humidity at a relative humidity of 100% is quite high relative to Antarctica.

AFA^{2}D

Definitions

- Psychrometrics

    The field of engineering concerned with the determination of physical and thermodynamic properties of gas-vapor mixtures.



AFA2D
Definitions
• Psychrometric Chart
Scary, complicated looking graph.
ALTITUDE: SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. HG
ATMOSPHERIC PRESSURE: 14.696 psia
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY - BTU PER POUND OF DRY AIR
RELATIVE HUMIDITY
DRY BULB TEMPERATURE - °F
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
ENTHALPY - BTU PER POUND OF DRY AIR
Chart by: AKTON PSYCHROMETRICS, www.aktonassoc.com

AFA^{2}D

Definitions

- Psychrometric Equations

  The alternative to using the psych chart.

$$v = \frac{1}{x_a}\left[\left|\frac{RT}{p}\right| \bullet \frac{1}{a} \bullet \left(x_a{}^2 A_{aa} + 2x_a x_w A_{aw} - X_a{}^3 A_{www}\, p\right)\beta\right]$$

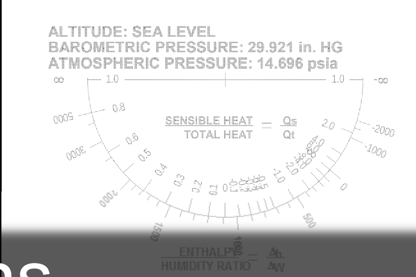
$$h = \left[x_a h_a{}^{\circ} + \left(0.62198 x_w h_w{}^{\circ}\right)\beta - \left(x_a{}^2 B_{aa} + 2x_a x_w B_{aw} + x_w{}^2 B_{aw} + x_w{}^2 B_{ww}\right)\bullet\, p\alpha - \frac{1}{2} x_w{}^3 B_{www}\, p^2 \alpha\right]\frac{1}{x_a} + \bar{h}_a W \bar{h}_w$$

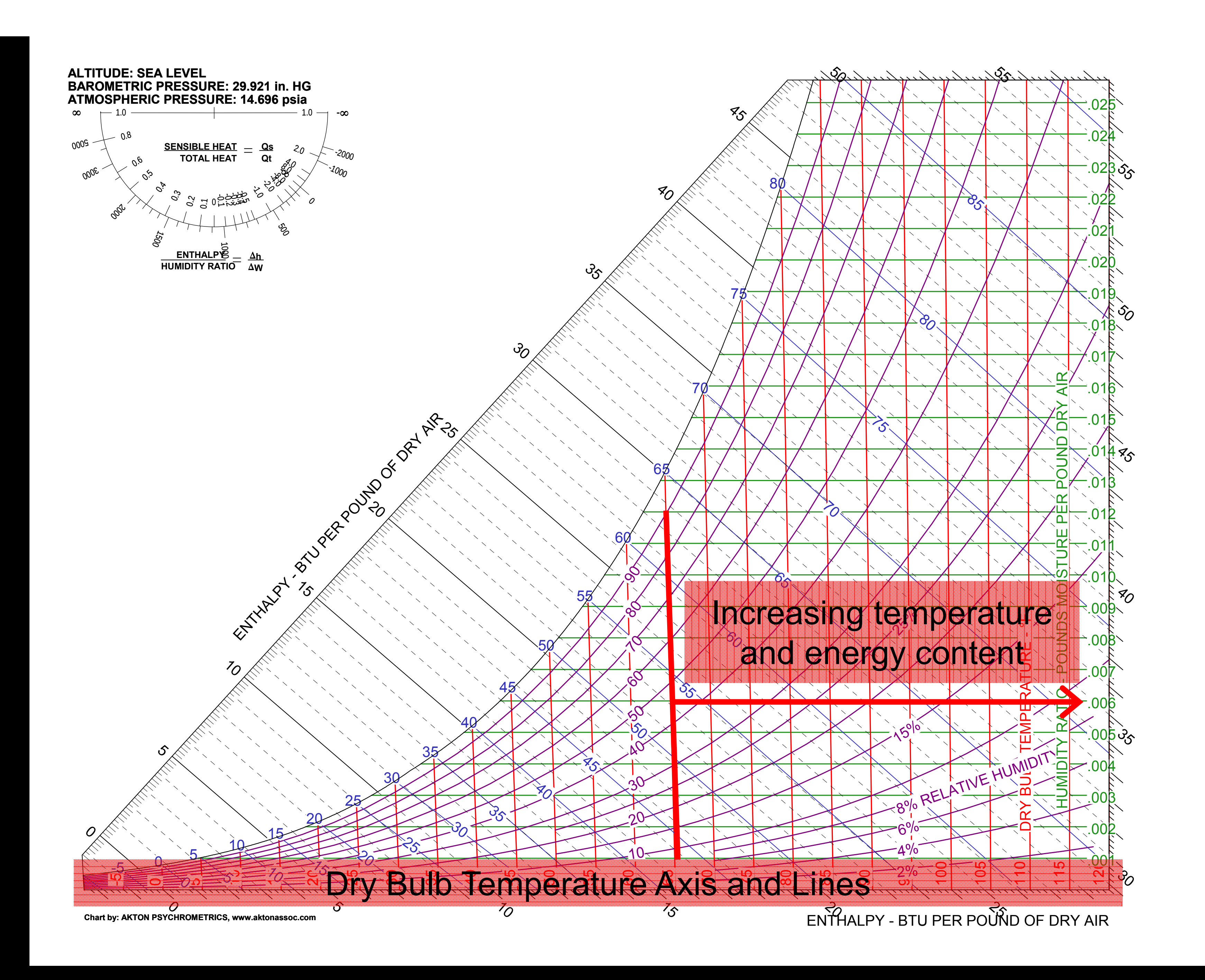
AFA^{2}D

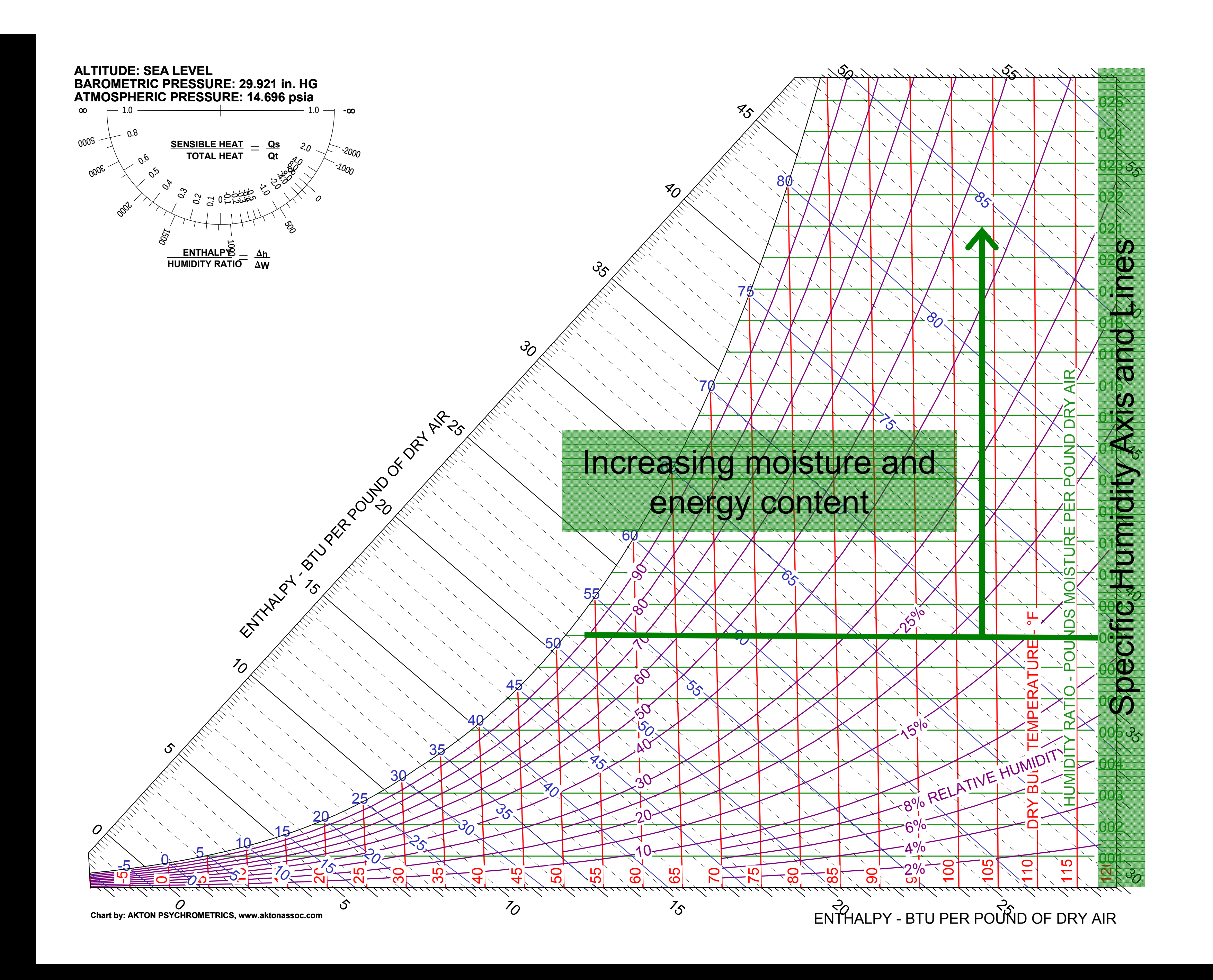
Definitions

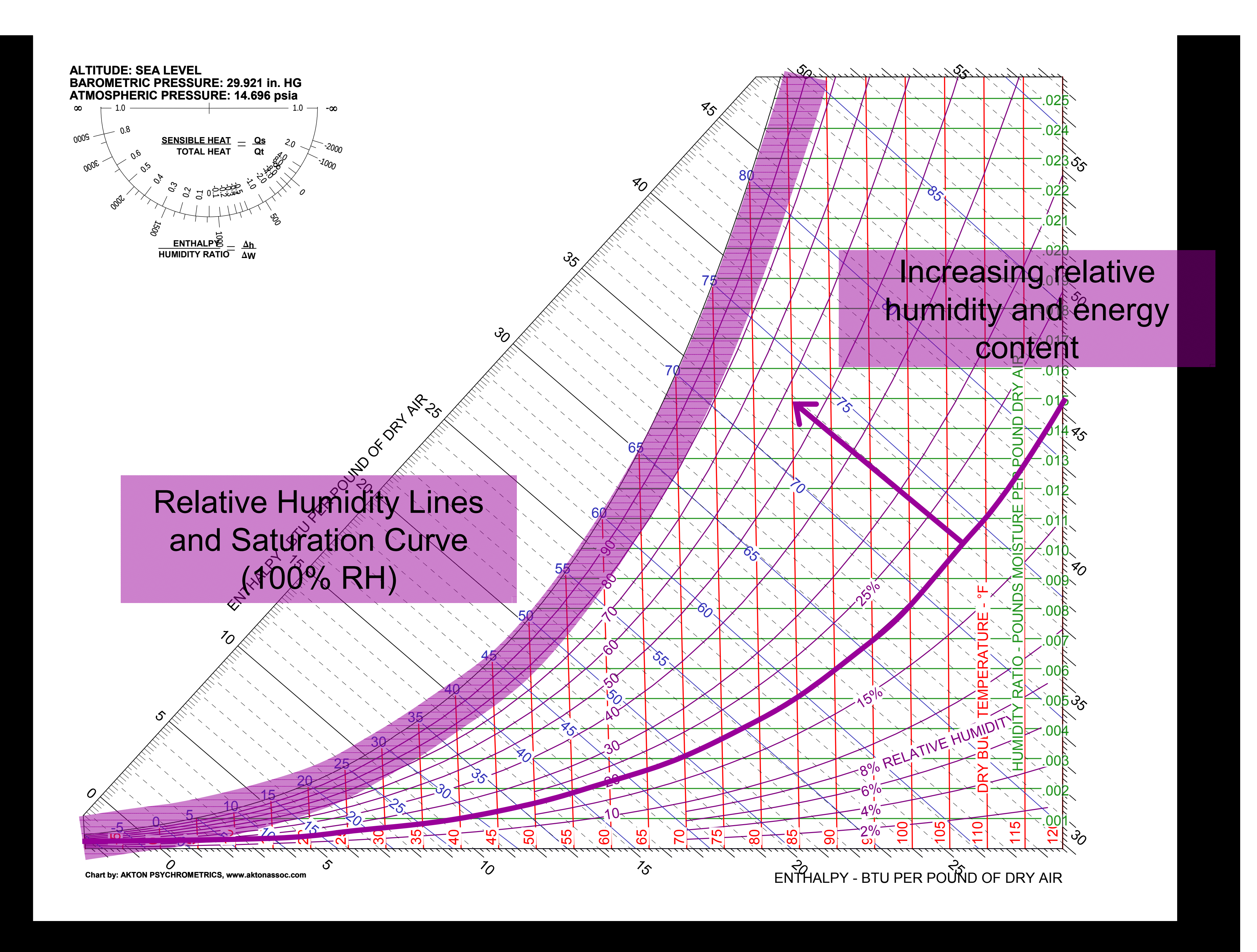
- Psychrometric Chart

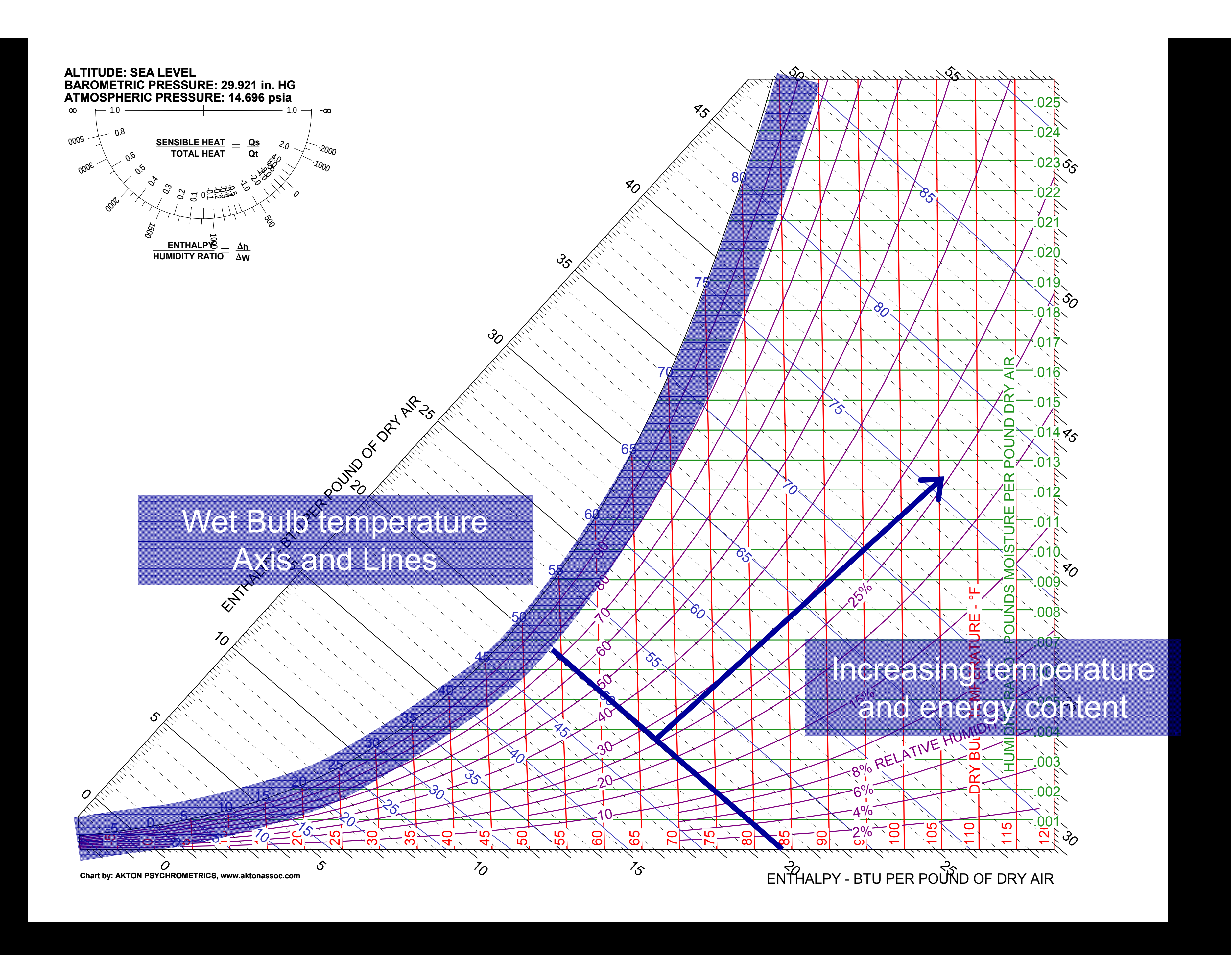
  Graphical tool that allows the informed user to determine multiple parameters like enthalpy, dew point, relative humidity, specific humidity, dry bulb temperature and wet bulb temperature for a sample of air if any two of them are know.

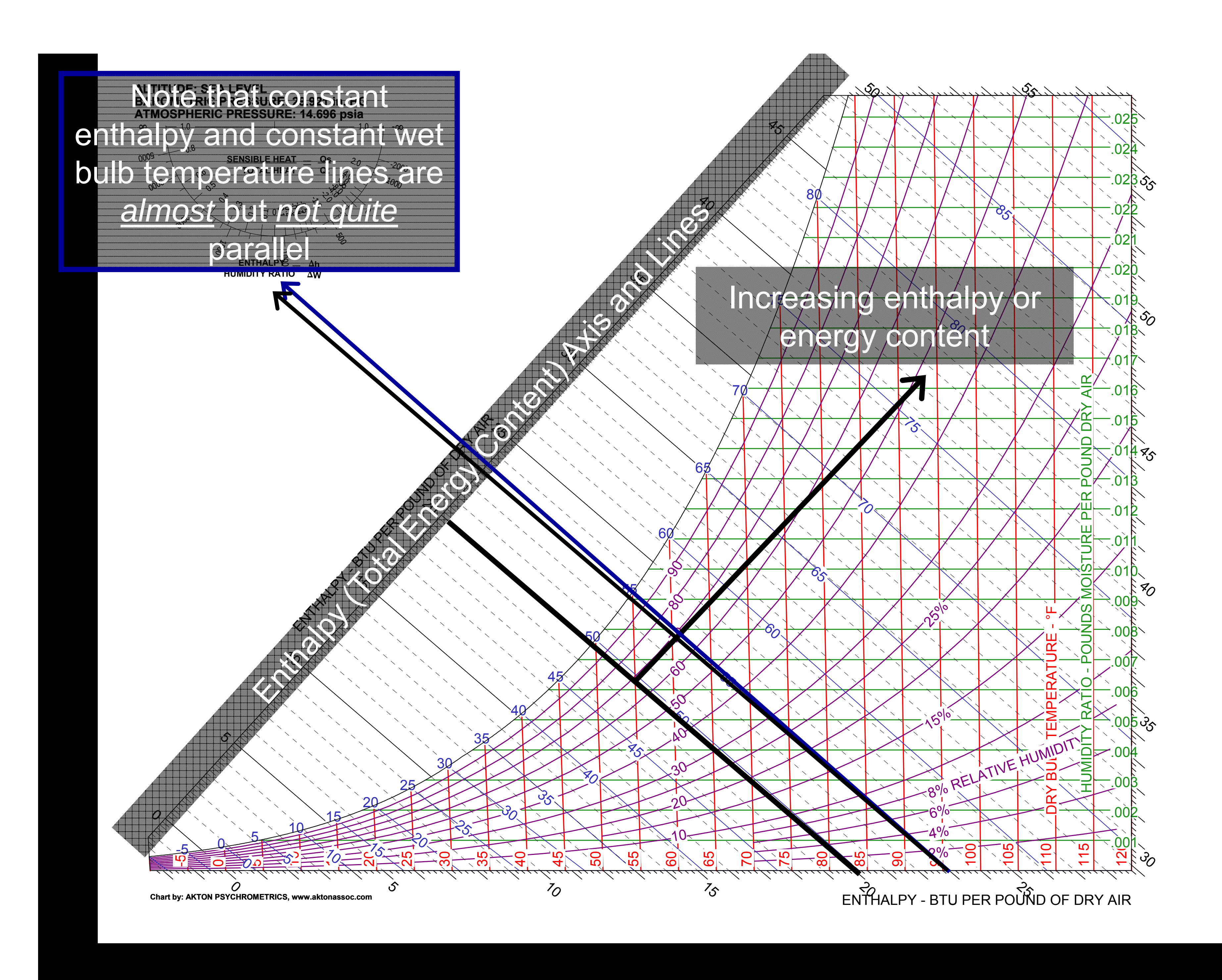



ALTITUDE: SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. HG
ATMOSPHERIC PRESSURE: 14.696 psia
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY - BTU PER POUND OF DRY AIR
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE
RELATIVE HUMIDITY
Increasing temperature and energy content
Dry Bulb Temperature Axis and Lines
Chart by: AKTON PSYCHROMETRICS, www.aktonassoc.com

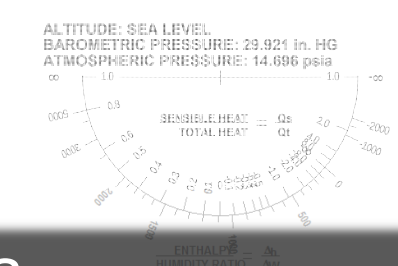

ALTITUDE: SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. HG
ATMOSPHERIC PRESSURE: 14.696 psia
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY - BTU PER POUND OF DRY AIR
Increasing moisture and energy content
Specific Humidity Axis and Lines
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE - °F
RELATIVE HUMIDITY
Chart by: AKTON PSYCHROMETRICS, www.aktonassoc.com
ENTHALPY - BTU PER POUND OF DRY AIR


ALTITUDE: SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. HG
ATMOSPHERIC PRESSURE: 14.696 psia
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
Relative Humidity Lines and Saturation Curve (100% RH)
Increasing relative humidity and energy content
ENTHALPY - BTU PER POUND OF DRY AIR
DRY BULB TEMPERATURE - °F
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
8% RELATIVE HUMIDITY
25%
15%
6%
4%
2%
Chart by: AKTON PSYCHROMETRICS, www.aktonassoc.com


ALTITUDE: SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. HG
ATMOSPHERIC PRESSURE: 14.696 psia
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
Wet Bulb temperature Axis and Lines
Increasing temperature and energy content
ENTHALPY - BTU PER POUND OF DRY AIR
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE - °F
RELATIVE HUMIDITY
Chart by: AKTON PSYCHROMETRICS, www.aktonassoc.com


Note that constant enthalpy and constant wet bulb temperature lines are *almost* but *not quite* parallel
Enthalpy (Total Energy Content) Axis and Lines
Increasing enthalpy or energy content
ENTHALPY - BTU PER POUND OF DRY AIR
DRY BULB TEMPERATURE - °F
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
RELATIVE HUMIDITY
Chart by: AKTON PSYCHROMETRICS, www.aktonassoc.com

AFA²D



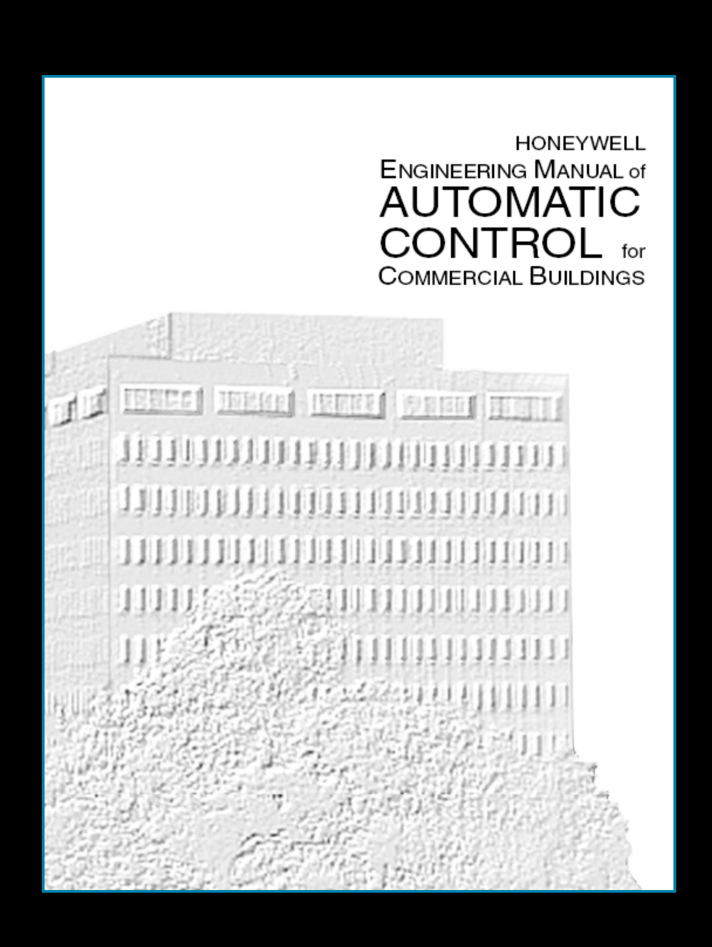
Definitions

- Psychrometric Chart

  If you know the dry bulb temperature and relative humidity entering and leaving a cooling coil, you can plot those points on the chart and determine:

  - The entering and leaving enthalpy (energy content)
  - The load (how much energy was transferred)
  - The amount of moisture condensed, if any
  - The entering and leaving dew point temperature
  - The entering and leaving wet bulb temperature

Learn More about Using a Psych Chart



HTML version at:

http://www.buildingcontrolworkbench.com

Downloadable .pdf at:

http://customer.honeywell.com/techlit/pdf/77-0000s/77-E1100.pdf

AFA^{2}D

Definitions

- Cooling

  A process that removes energy. For a space, this is often accomplished by circulating air through it at a temperature below the required set point. For an airstream, this is often accomplished by passing it over a surface that is below the required supply temperature. If the surface is below the dew point of the air stream, dehumidification (moisture removal) will also occur.

AFA^{2}D

Definitions

- Heating

A process that adds energy. For a space, this is often accomplished by circulating air through it at a temperature above the required set point. For an airstream, this is often accomplished by passing it over a surface that is above the required supply temperature.

AFA^{2}D

Definitions

- Freezing

    A condition that occurs when water is cooled to the point where it changes phase from a solid to a liquid.

AFA^{2}D

Definitions

- Water Damage

  A condition that occurs after frozen water contained in a HVAC coil changes back to the liquid phase.

AFA²D

Definitions

- Expletive

  A generic reference to the field terminology used to describe and discuss water damage when it occurs.

AFA^{2}D

Definitions

- Preheat

A process that heats a fluid stream to prepare it for a subsequent HVAC process. In air handling systems, this process is used to raise subfreezing air above freezing to protect water filled elements down stream from damage due to freezing.

See the Functional Testing Guide (www.peci.org/ftguide) Air Handling System Reference Guide Chapter 5 – Preheat, Table 5.1 to contrast preheat, reheat and heating applications

AFA^{2}D

Definitions

- Reheat

  A process that uses heat to warm air being delivered to a zone to prevent over cooling. The temperature of the air was set by the need to hit a dehumidification target or by the requirements of another zone, so it can not be raised at the central system. The volume can not be reduced because it has been set to assure proper ventilation (contaminant control). In the limit, reheat will raise the supply temperature to the zone temperature but not above it.

AFA^{2}D

Definitions

- Economizer Process

  An HVAC process designed to minimize the energy required to cool a building

AFA^{2}D

Definitions

- Constant Volume System

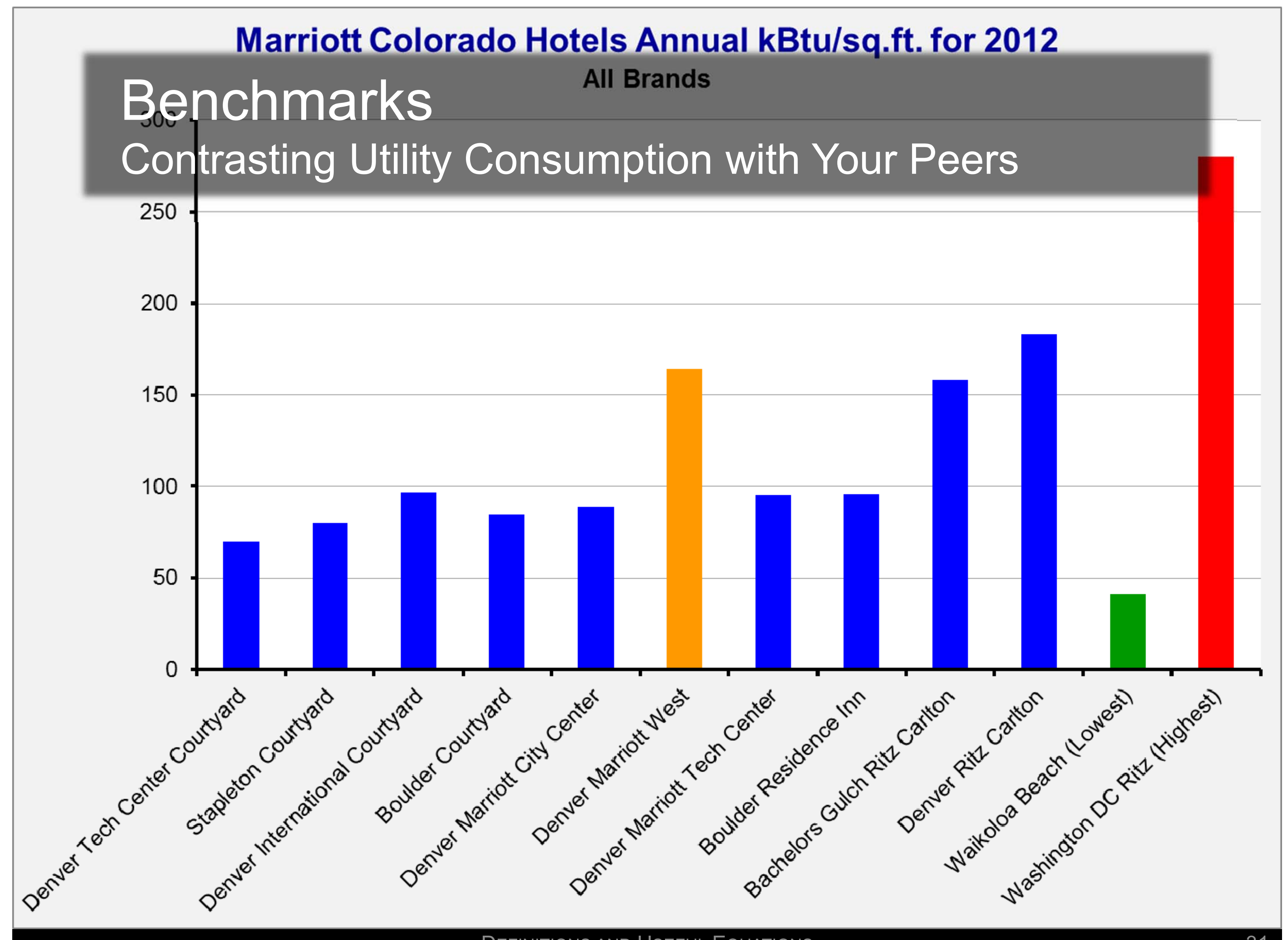
  An air handling or pumping process that, in general terms, is always moving the same amount of water or air. Pump or fan energy is fairly steady state. Supply and return temperature differences will tend to vary with load. In water systems, the control valves will tend to be three-way valves.

AFA^{2}D

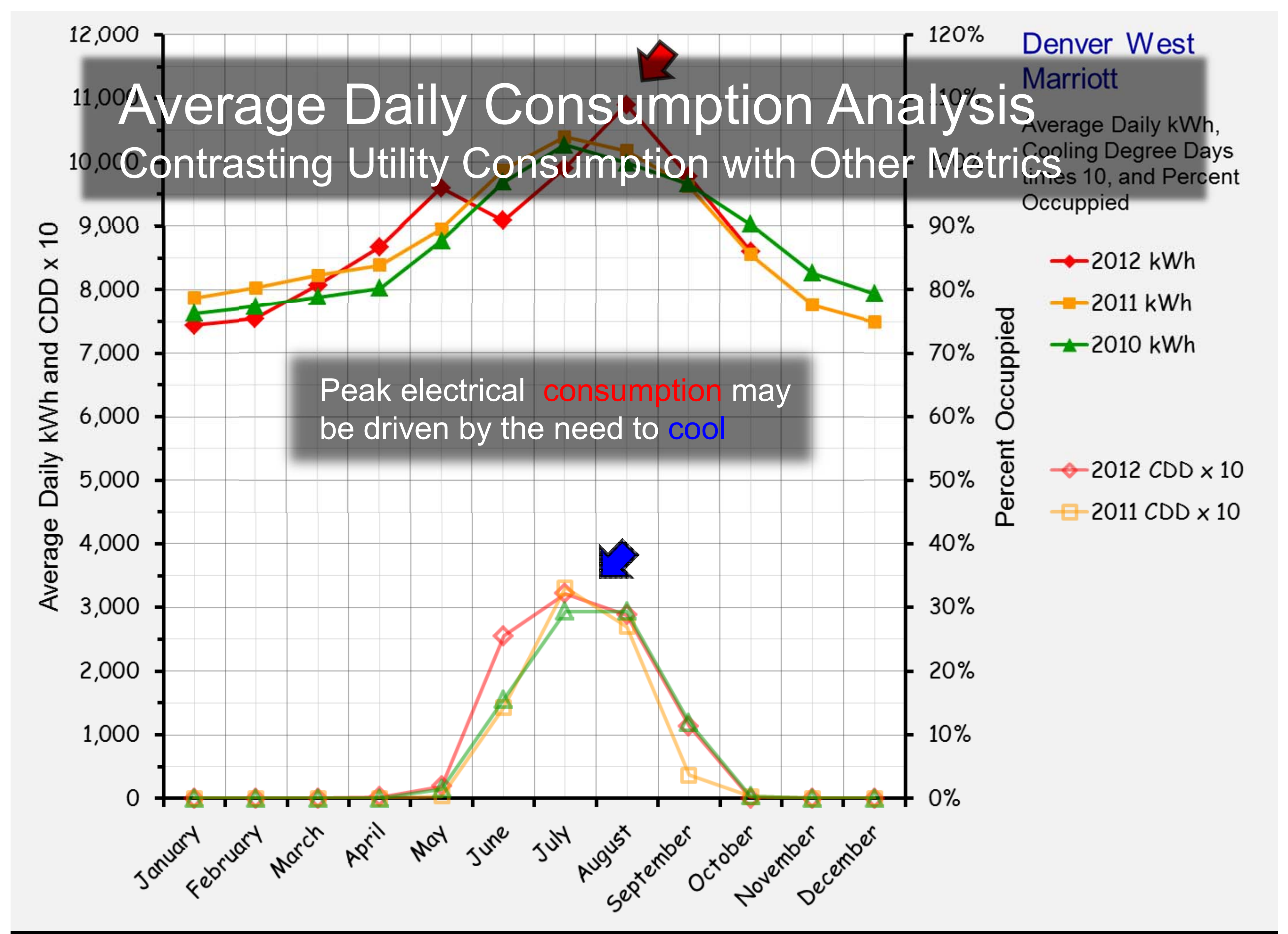
Definitions

- Variable Volume System/Variable Air Volume System (VAV)

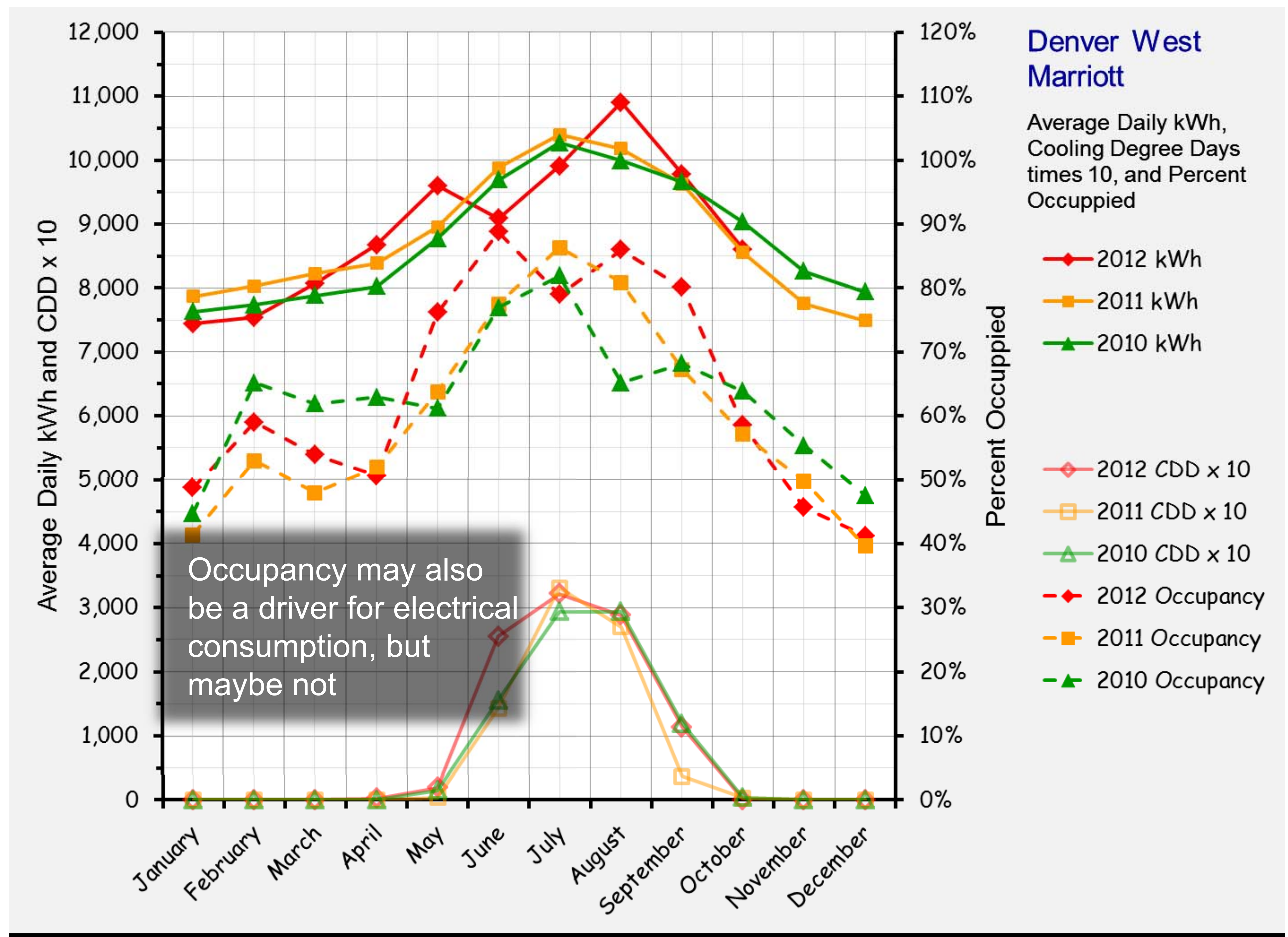
  An air handling or pumping process that varies the flow of water or air to match the requirements of the load.. Supply and return temperature differences will tend to hold steady regardless of load. In water systems, the control valves will tend to be two-way valves.

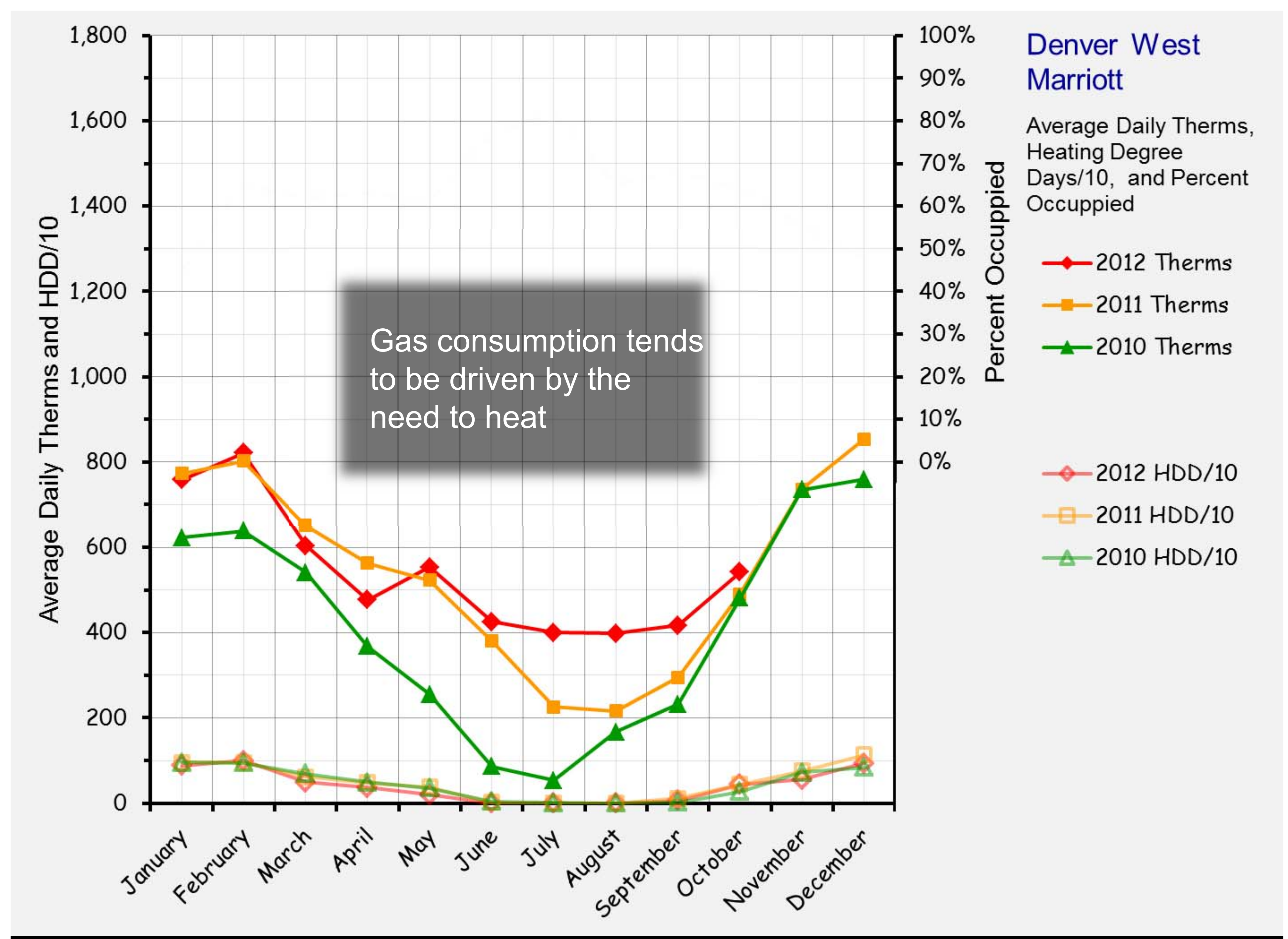


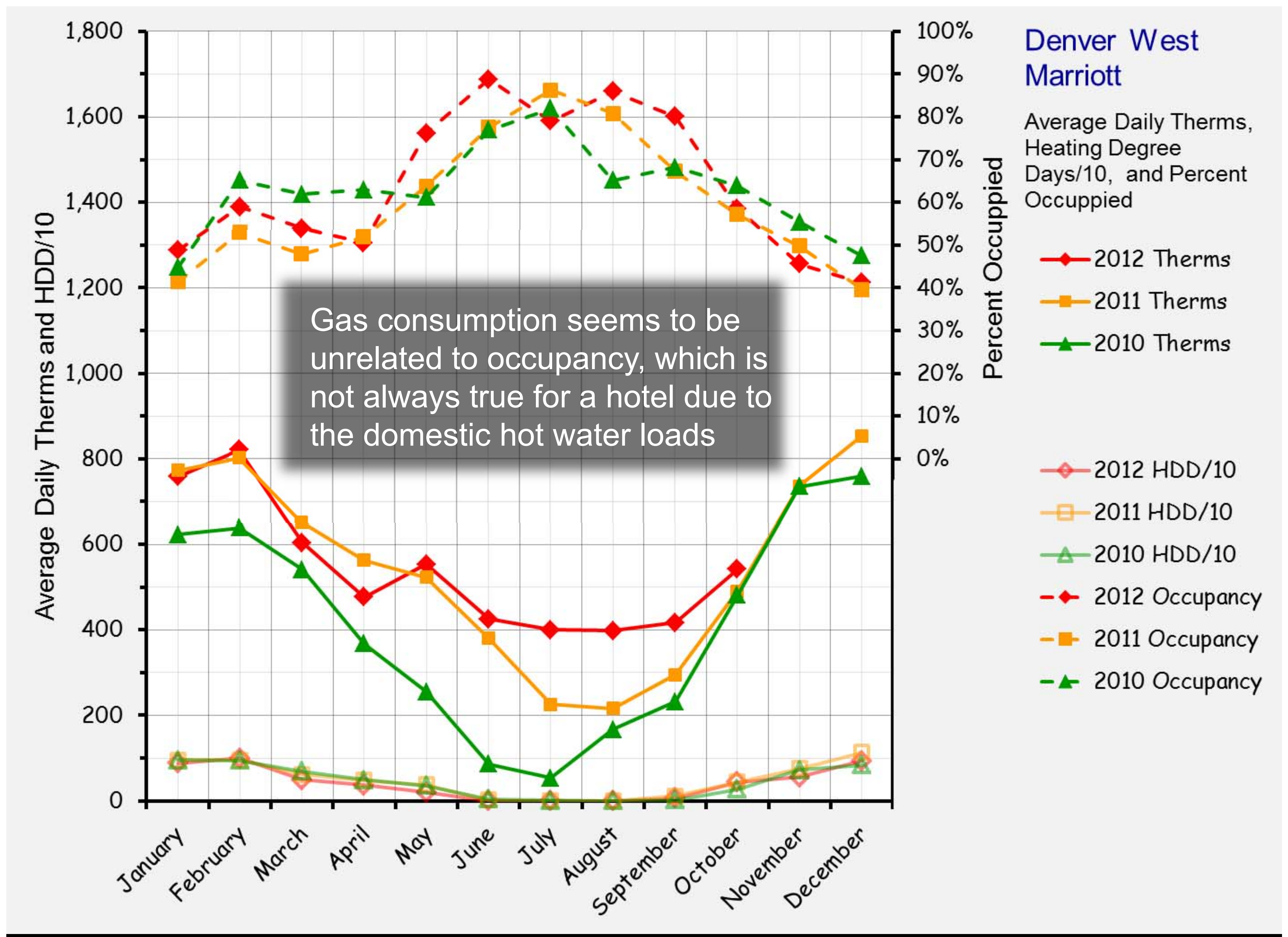
Marriott Colorado Hotels Annual kBtu/sq.ft. for 2012
All Brands
Benchmarks
Contrasting Utility Consumption with Your Peers
250
200
150
100
50
0
Denver Tech Center Courtyard
Stapleton Courtyard
Denver International Courtyard
Boulder Courtyard
Denver Marriott City Center
Denver Marriott West
Denver Marriott Tech Center
Boulder Residence Inn
Bachelors Gulch Ritz Carlton
Denver Ritz Carlton
Waikoloa Beach (Lowest)
Washington DC Ritz (Highest)



Average Daily Consumption Analysis
Contrasting Utility Consumption with Other Metrics
Denver West Marriott
Average Daily kWh, Cooling Degree Days times 10, and Percent Occuppied
Peak electrical consumption may be driven by the need to cool
Average Daily kWh and CDD x 10
Percent Occuppied
2012 kWh
2011 kWh
2010 kWh
2012 CDD x 10
2011 CDD x 10
January
February
March
April
May
June
July
August
September
October
November
December


Denver West Marriott
Average Daily kWh, Cooling Degree Days times 10, and Percent Occuppied
Average Daily kWh and CDD x 10
Percent Occuppied
12,000
11,000
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
0
120%
110%
100%
90%
80%
70%
60%
50%
40%
30%
20%
10%
0%
January
February
March
April
May
June
July
August
September
October
November
December
2012 kWh
2011 kWh
2010 kWh
2012 CDD x 10
2011 CDD x 10
2010 CDD x 10
2012 Occupancy
2011 Occupancy
2010 Occupancy
Occupancy may also be a driver for electrical consumption, but maybe not


Denver West Marriott
Average Daily Therms, Heating Degree Days/10, and Percent Occuppied
Gas consumption tends to be driven by the need to heat
Average Daily Therms and HDD/10
Percent Occuppied
1,800
1,600
1,400
1,200
1,000
800
600
400
200
0
100%
90%
80%
70%
60%
50%
40%
30%
20%
10%
0%
2012 Therms
2011 Therms
2010 Therms
2012 HDD/10
2011 HDD/10
2010 HDD/10
January
February
March
April
May
June
July
August
September
October
November
December


Denver West Marriott
Average Daily Therms, Heating Degree Days/10, and Percent Occuppied
Gas consumption seems to be unrelated to occupancy, which is not always true for a hotel due to the domestic hot water loads
Average Daily Therms and HDD/10
Percent Occuppied
1,800
1,600
1,400
1,200
1,000
800
600
400
200
0
100%
90%
80%
70%
60%
50%
40%
30%
20%
10%
0%
January
February
March
April
May
June
July
August
September
October
November
December
2012 Therms
2011 Therms
2010 Therms
2012 HDD/10
2011 HDD/10
2010 HDD/10
2012 Occupancy
2011 Occupancy
2010 Occupancy

Interval Data Consumption Analysis

Looking at Variations in Hour by Hour Patterns

Typical Day, Hour by Hour kW Profile, 2009 - 2012

2012, Debugged

kW

150 140 130 120 110 100 90 80 70 60 50 40 30 20 10 0

00 01 02 03 04 05 06 07 08 09 10 11 12 13 14 15 16 17 18 19 20 21 22 23 00

Sensible Heating or Cooling Loads

$$Q = 1.08 \times cfm \times \Delta t$$

Where :

Q = Load in btu/hr

1.08 = A units conversion constant

cfm = Flow rate in cubic feet per minute

Δt = Temperature difference across the element in °F

Where did the Units Conversion Constant Come From?

$$\frac{1 \text{ btu}}{\text{hour}} = \frac{1 \text{ cubic ft.}}{\text{minute}}$$

Where did the Units Conversion Constant Come From?

$$\frac{1 \text{ btu}}{\text{hour}} = \frac{1 \ \cancel{\text{cubic ft.}}}{\cancel{\text{minute}}} \times \frac{60 \ \cancel{\text{minutes}}}{\text{hour}} \times \frac{0.0749 \ \cancel{\text{pounds}}}{\cancel{\text{cubic ft.}}} \times \frac{0.240 \text{ btu}}{\cancel{\text{pound}} \ \cancel{^\circ\text{F}}} \times 1 \ \cancel{^\circ\text{F}}$$

$$\frac{1 \text{ btu}}{\text{hour}} = \frac{1.08 \text{ cubic ft.}}{\text{minute}} \times \Delta t, ^\circ\text{F}$$

Physical Properties can Vary …

Specific Heat, btu/lb-°F

0.260
0.250
0.240
0.230
0.220
0.210
0.200

Density, lb/cubic foot

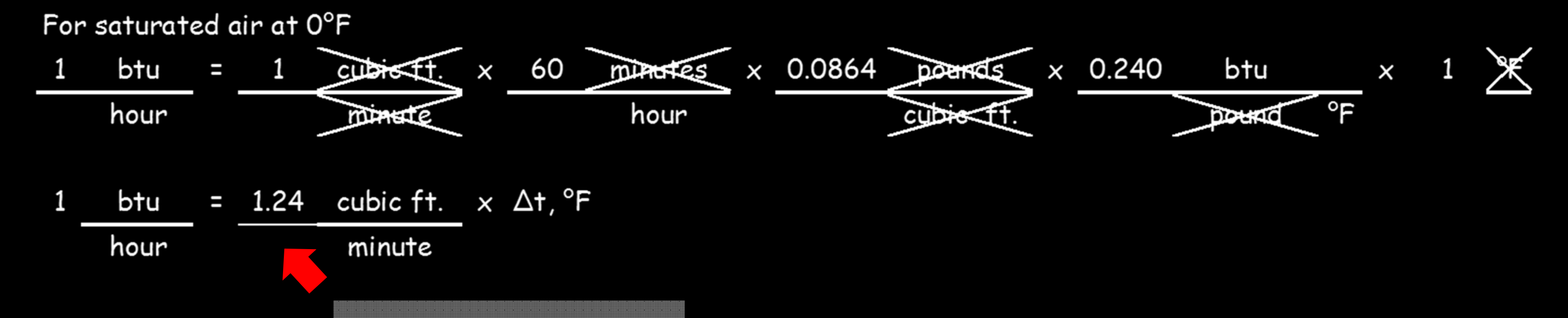
0.24
0.20
0.16
0.12
0.08
0.04
0.00

-100 0 100 200 300 400 500

Temperature,°F

Specific Heat
Density of Dry Air
Density of Saturated Air

… so Conversion Constants Only are Valid for a Range of Conditions



Versus 1.08 in the equation in common use

… so Conversion Constants Only are Valid for a Range of Conditions

For saturated air at 0°F

$$\frac{1\ \text{btu}}{\text{hour}} = \frac{1\ \cancel{\text{cubic ft.}}}{\cancel{\text{minute}}} \times \frac{60\ \cancel{\text{minutes}}}{\text{hour}} \times \frac{0.0864\ \cancel{\text{pounds}}}{\cancel{\text{cubic ft.}}} \times \frac{0.240\ \text{btu}}{\cancel{\text{pound}}\ {}^\circ\text{F}} \times 1\ \cancel{{}^\circ\text{F}}$$

$$1\ \frac{\text{btu}}{\text{hour}} = 1.24\ \frac{\text{cubic ft.}}{\text{minute}} \times \Delta t,\ {}^\circ\text{F}$$

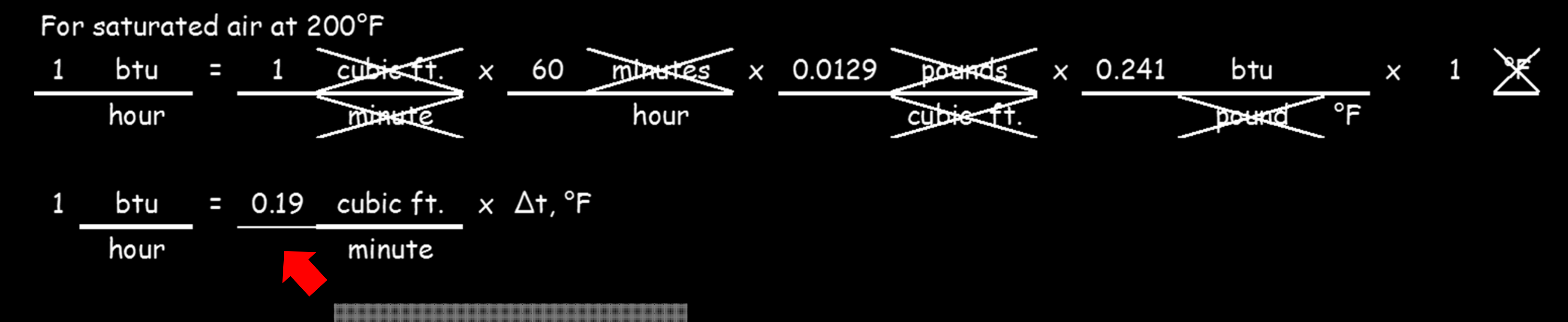
For dry air at 0°F

$$\frac{1\ \text{btu}}{\text{hour}} = \frac{1\ \cancel{\text{cubic ft.}}}{\cancel{\text{minute}}} \times \frac{60\ \cancel{\text{minutes}}}{\text{hour}} \times \frac{0.0864\ \cancel{\text{pounds}}}{\cancel{\text{cubic ft.}}} \times \frac{0.240\ \text{btu}}{\cancel{\text{pound}}\ {}^\circ\text{F}} \times 1\ \cancel{{}^\circ\text{F}}$$

$$1\ \frac{\text{btu}}{\text{hour}} = 1.24\ \frac{\text{cubic ft.}}{\text{minute}} \times \Delta t,\ {}^\circ\text{F}$$

Virtually the same as for *cold* dry air

… so Conversion Constants Only are Valid for a Range of Conditions



Versus 1.08 in the equation in common use

… so Conversion Constants Only are Valid for a Range of Conditions

For saturated air at 200°F

$$\frac{1 \text{ btu}}{\text{hour}} = \frac{1 \text{ cubic ft.}}{\text{minute}} \times \frac{60 \text{ minutes}}{\text{hour}} \times \frac{0.0129 \text{ pounds}}{\text{cubic ft.}} \times \frac{0.241 \text{ btu}}{\text{pound } ^\circ\text{F}} \times 1\ ^\circ\text{F}$$

$$\frac{1 \text{ btu}}{\text{hour}} = 0.19 \frac{\text{cubic ft.}}{\text{minute}} \times \Delta t, ^\circ\text{F}$$

For dry air at 200°F

$$\frac{1 \text{ btu}}{\text{hour}} = \frac{1 \text{ cubic ft.}}{\text{minute}} \times \frac{60 \text{ minutes}}{\text{hour}} \times \frac{0.06 \text{ pounds}}{\text{cubic ft.}} \times \frac{0.241 \text{ btu}}{\text{pound } ^\circ\text{F}} \times 1\ ^\circ\text{F}$$

$$\frac{1 \text{ btu}}{\text{hour}} = 0.87 \frac{\text{cubic ft.}}{\text{minute}} \times \Delta t, ^\circ\text{F}$$

Significantly different from hot saturated air and the value in the equation in common

Latent Load

$$Q = .68 \times cfm \times \Delta w$$

Where :

Q = Load in btu/hr

.68 = A units conversion constant

cfm = Flow rate in cubic feet per minute

Δw = Specific humidity change in grains of moisture per pound of dry air.

(there are 7,000 grains per pound)

Total Load

$$Q = 4.5 \times cfm \times \Delta h$$

Where :

Q = Load in btu/hr

4.5 = A units conversion constant

cfm = Flow rate in cubic feet per minute

Δh = Enthalpy difference across the element in Btu/lb

Water Side Load

$$Q = 500 \times gpm \times \Delta t$$

Where :

Q = Load in btu/hr

500 = A units conversion constant

gpm = Flow rate in gallons per minute

Δt = Temperature difference across the element in °F

The Relationship Between Flow and Velocity

$$Q = VA$$

Where:

Q = Flow rate in cubic feet per minute

V = Velocity in feet per minute

A = Cross sectional area in square feet

The Relationship Between Velocity and Velocity Pressure

$$V = 4{,}005\sqrt{VP}$$

Where :

V = Velocity in feet per minute

4,005 = A units conversion constant

VP = Velocity pressure in inches w.c.

Fan Power

$$bhp = \left(\frac{cfm \times static}{6{,}356 \times \eta_{fan_{static}}} \right)$$

Where :

bhp = Brake horse power into the fan drive shaft

cfm = Flow rate in cubic feet per minute

$static$ = Fan static pressure

$6{,}356$ = A units conversion constant

$\eta_{fan_{static}}$ = Fan static efficiency; .40 = .60 for small fans,

.68 - .78 for large fans

Divide by motor efficiency and multiply by .746 kW

per horse power to get killoWatts

Unit Conversions for Working with SI Units

$$1 \text{ cubic foot per minute (cfm)} = 0.02831684659 \text{ cubic meters per hour } (m^3/hr)$$

$$1 \text{ cubic foot per minute (cfm)} = 7.86579072 \times 10^{-6} \text{ cubic meters per second } (m^3/sec)$$

$$1 \text{ inch of static (in.w.c. or in.} H_2O) = 0.2490889 \text{ killopascals of static (kPa)}$$

Pump Power

$$bhp = \left(\frac{gpm \times head}{3{,}960 \times \eta_{pump}} \right)$$

Where :

bhp = Brake horse power into the pump drive shaft

gpm = Flow rate in gallons per minute

$head$ = Pump head in feet water column

3,960 = A units conversion constant

η_{pump} = Pump efficiency; .40 - .70 for small (under 500 gpm) pumps, .70 - .85 for large pumps

Unit Conversions for Working with SI Units

$$1 \text{ gallon per minute (gpm)} = 0.227124707 \text{ cubic meters per hour } (m^3/hr)$$

$$1 \text{ gallon per minute (gpm)} = 6.30901964 \times 10^{-5} \text{ cubic meters per second } (m^3/sec)$$

$$1 \text{ foot of head (ft.w.c. or ft.} H_2O) = 0.3048 \text{ meters of head } (m\, H_2O)$$

Calculating Kw Into the Pump Motor

$$kW = \left(\frac{Flow_{gpm} \times Head_{ft.w.c.}}{3{,}960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{VSD}} \right) \times .746$$

Where:

kW = Input to the system to produce the flow and head.

Flow = Flow rate in gallons per minute. Generally speaking, we try to use a pump test for at least one condition as a basis for this. If that is not available we will use a value from a tab report. Lacking that we will use a design metric from the original drawings or an equipment submittal.

Head = The pump head in ft.w.c. water column, which we usually try to identify from field measurements and pump tests. Lacking those measurements we will use a value derived from a TAB report or the design value.

3,960 = A units conversion constant that is good for water between 40°F and 220°F.

η_{Pump} = Pump efficiency. We usually try to get this number from the pump curve or from the pump's rated brake horse power (bhp), flow and head. Lacking that, we will make a geometrically similar pump selection (same flow rate, head, impeller diameter, and speed) using manufacturer's software and use that efficiency. Lacking that is is reasonable to assume that for a pump rated for 300 gpm or less the efficiency might be in the range of 45-60%. For pumps rated between 300 gpm and 1,500 gpm, efficiencys might range from 60% to 75%. For pumps over 1,500 gpm, efficiencies might range from 75% to as high as 87%. Generally, efficiency will improve with pump size.

η_{Motor} = Motor efficiency. We usually try to get the motor performance curve and select the efficiency from the curve for the bhp that the pump impeller is extracting from it. If we can't get the motor curve, we use a similar motor selected from MotorMaster^{TM} International. In all cases we adjust the efficiency for the motor operating point vs. using the motor's rated nameplate efficiency. Lacking anything else, it is reasonable to assume that the motor efficiency will improve by 1-2% over the nameplate efficiency when the pump is at 65-85% of its rated load, drop back to near nameplate efficiency at around 50% load, and then drop sharply towards 0 at 20-30% of rated load.

Calculating Kw Into the Pump Motor (Continued)

$$kW = \left(\frac{Flow_{gpm} \times Head_{ft.w.c.}}{3,960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{VSD}} \right) \times .746$$

Where:

η_{VSD} = Variable speed drive efficiency. Where possible, we try to get the manufacturer's data for this. But this data is difficult to obtain and not consistent in its development. Lacking manufacture specific data, we use generic data as published by the Department of Energy on their Industrial Best Practices web site. Lacking any other source, it is reasonable to assume there will be at least 4-6% loss in the drive with it at full speed with a gradual decay to 80% efficiency at about 20% load.

.746 = Horsepower to kW conversion constant; there are .746 hp per kW, or stated mathematically with the appropriate units:

$$.746 \frac{kW}{\cancel{hp}} \times 1 \ \cancel{hp} = .746 \ kW$$

.

Calculating Power Into the Fan Motor as kW

$$kW = \left(\frac{Flow \times Static}{6{,}356 \times \ \eta_{Fan} \times \ \eta_{Motor} \times \ \eta_{Drive}} \right) \times .746$$

Where:

$kW =$ Electrical energy into the drive system seving the fan

$Flow =$ Flow rate in cubic feet per minute

$Static =$ Fan static in inches water column

6,356 = A units conversion constant

η_{Fan} = Fan efficiency

η_{Motor} = Motor efficiency

η_{Drive} = Drive efficiency; Don;'t forget about the belts if the motor is not direct drive. Well adjusted belts are 97-98% efficient. Poorly adjusted ones can be as low as 90% or less

746 = Horsepower to kW conversion constant; there are .746 hp per kW, or stated mathematically with the appropriate units:

$$.746 \frac{\text{kW}}{\cancel{\text{hp}}} \times 1 \ \cancel{\text{hp}} = .746 \ \text{kW}$$

Calculating Energy Use

$$kWh = \left(\frac{Flow \times Head}{3{,}960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{Drive}} \right) \times \frac{.746\ \text{kw}}{\text{hp}} \times hours$$

Where:

$hours$ = Hours at the load condition defined by the other parameters

What can cause the load to vary?

- Changes in ambient conditions
- Changes in internal conditions
- Changes in a production process

Just About Everything!

The Square Law

$$P_{New} = P_{Old} \times \left(\frac{Flow_{New}}{Flow_{Old}} \right)^2$$

Where :

P_{New} = Pressure drop through a system at a new flow rate in consistent units

P_{Old} = Pressure drop through a system at a known flow rate in consistent units

$Flow_{New}$ = Flow rate for which the new pressure drop is to be calculated in consistent units

$Flow_{Old}$ = Known flow rate that produced the known pressure drop in consistent units

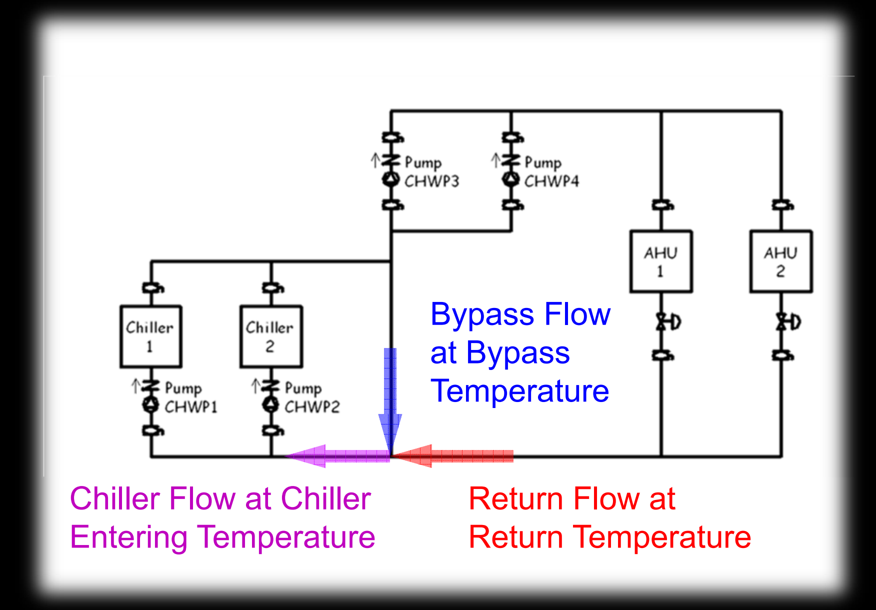
Conservation of Mass and Energy

The Goes Inta's gotta equal the Goes Outa's

Dr. Al Black

The tee where the building return meets the bypass line is a node in the system

- Energy into and out of the node must be equal
- Mass flow into and out of the node must be equal



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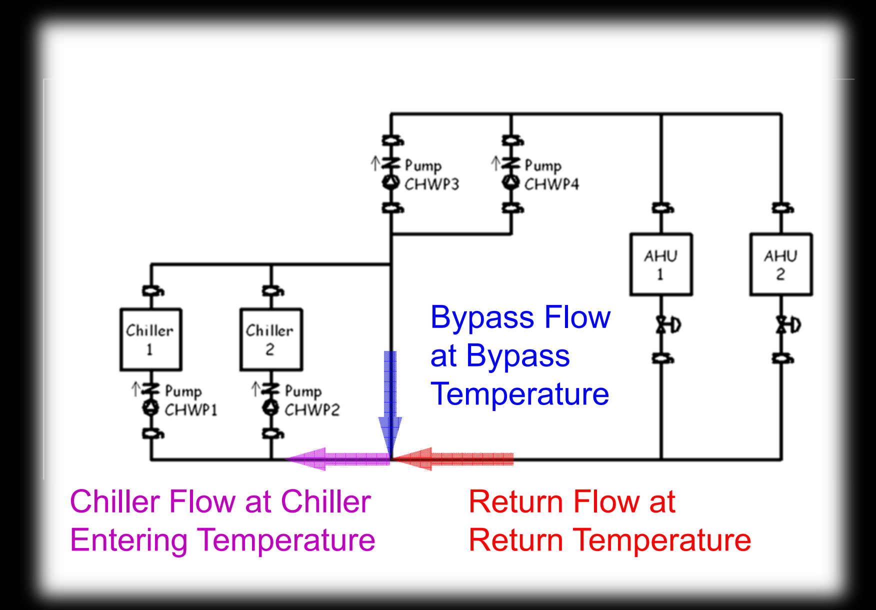
- Mass flow into and out of the node must be equal
- This is a steady state, steady flow process described by the continuity equation



$$q + h_i + \frac{V_i^2}{2g_c} + Z_i \frac{g}{g_c} = h_e + \frac{V_e^2}{2g_c} + Z_e \frac{g}{g_c} + w$$

Where:

q = Heat transfer — h = Enthalpy

V = Velocity — Z = Elevation

w = Work — g = Acceleration due to gravity

g_c = Constant relating force, mass, length and time

$_i$ Subscript = Incoming or entering conditions

$_e$ Subscript = Existing or leaving conditions

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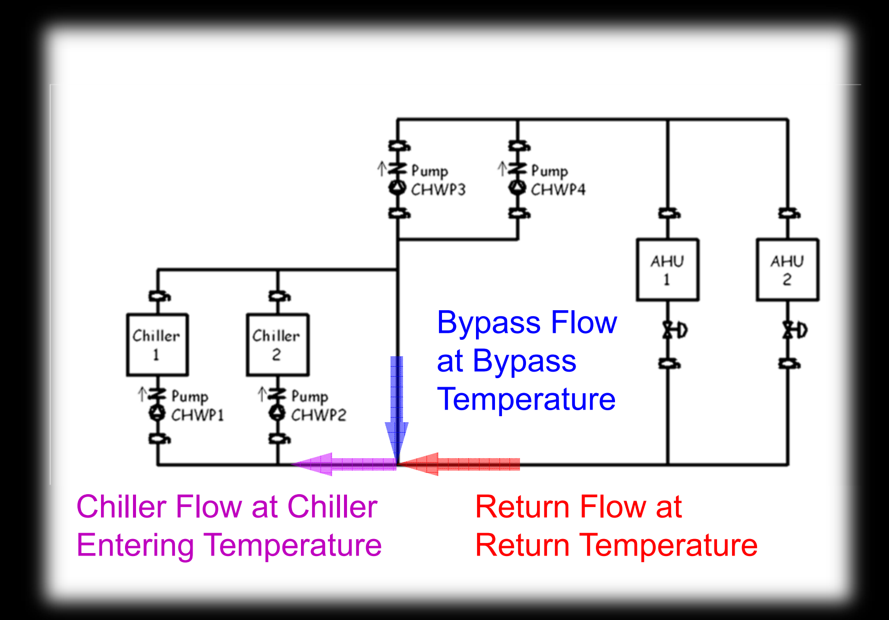
- Energy into and out of the node must be equal
- Mass flow into and out of the node must be equal
- This is a steady state, steady flow process described by the continuity equation
- For a tee in the pipe, the continuity equation can be simplified



$$\frac{\left(Flow_{Return} \times Temperature_{Return}\right) + \left(Flow_{Bypass} \times Temperature_{Bypass}\right)}{\left(Flow_{Chiller} \times Temperature_{Chiller}\right)}$$

Conservation of Mass and Energy

Doing the Algebra

$$\left(Flow_{Return} \times Temperature_{Return}\right) + \left(Flow_{Bypass} \times Temperature_{Bypass}\right) = \left(Flow_{Chiller} \times Temperature_{Chiller}\right)$$

Conservation of Mass and Energy

Doing the Algebra

$$\left(Flow_{Return} \times Temperature_{Return}\right) + \left(Flow_{Bypass} \times Temperature_{Bypass}\right) = \left(Flow_{Chiller} \times Temperature_{Chiller}\right)$$

Isolating the bypass parameters to one side of the equation:

$$\left(Flow_{Bypass} \times Temperature_{Bypass}\right) = \left(Flow_{Chiller} \times Temperature_{Chiller}\right) - \left(Flow_{Return} \times Temperature_{Return}\right)$$

Conservation of Mass and Energy

Doing the Algebra

$$(Flow_{Return} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller})$$

Isolating the bypass parameters to one side of the equation:

$$(Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller}) - (Flow_{Return} \times Temperature_{Return})$$

The return flow can be expressed in terms of the bypass flow and chiller flow as follows:

$$Flow_{Return} = Flow_{Chiller} - Flow_{Bypass}$$

Conservation of Mass and Energy

Doing the Algebra

$$(Flow_{Return} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller})$$

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$$Flow_{Return} = Flow_{Chiller} - Flow_{Bypass}$$

Substituting the new equation for return flow into the original equation:

$$(Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller}) - ((Flow_{Chiller} - Flow_{Bypass}) \times Temperature_{Return})$$

Conservation of Mass and Energy

Doing the Algebra

$$(Flow_{Return} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller})$$

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Eliminating one set of parenthesis:

$$(Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller}) - (Flow_{Chiller} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Return})$$

Conservation of Mass and Energy

Doing the Algebra

$$(Flow_{Return} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller})$$

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Eliminating one set of parenthesis:

$$(Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller}) - (Flow_{Chiller} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Return})$$

Rearranging terms:

$$(Flow_{Bypass} \times Temperature_{Bypass}) - (Flow_{Bypass} \times Temperature_{Return}) = (Flow_{Chiller} \times Temperature_{Chiller}) - (Flow_{Chiller} \times Temperature_{Return})$$

Conservation of Mass and Energy

Doing the Algebra

$$(Flow_{Return} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller})$$

Isolating the bypass parameters to one side of the equation:

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Eliminating one set of parenthesis:

$$(Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller}) - (Flow_{Chiller} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Return})$$

Rearranging terms:

$$(Flow_{Bypass} \times Temperature_{Bypass}) - (Flow_{Bypass} \times Temperature_{Return}) = (Flow_{Chiller} \times Temperature_{Chiller}) - (Flow_{Chiller} \times Temperature_{Return})$$

Eliminating two sets of parenthesis:

$$Flow_{Bypass} \times (Temperature_{Bypass} - Temperature_{Return}) = Flow_{Chiller} \times (Temperature_{Chiller} - Temperature_{Return})$$

Conservation of Mass and Energy

Doing the Algebra

$$(Flow_{Return} \times Temperature_{Return}) + (Flow_{Bypass} \times Temperature_{Bypass}) = (Flow_{Chiller} \times Temperature_{Chiller})$$

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Rearranging terms:

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Eliminating two sets of parenthesis:

$$Flow_{Bypass} \times (Temperature_{Bypass} - Temperature_{Return}) = Flow_{Chiller} \times (Temperature_{Chiller} - Temperature_{Return})$$

Solving for Bypass Flow $(Flow_{Bypass})$:

$$Flow_{Bypass} = \frac{Flow_{Chiller} \times (Temperature_{Chiller} - Temperature_{Return})}{(Temperature_{Bypass} - Temperature_{Return})}$$

Conservation of Mass and Energy
Doing the Algebra

Because :

$$\%_{Bypass\ Flow} + \%_{Return\ Flow} = 100\%_{Chiller\ Flow}$$

You can solve for bypass flow as a percentage of total flow to the chiller as follows :

$$Flow_{Bypass} = \frac{1 \times \left(Temperature_{Chiller} - Temperature_{Return}\right)}{\left(Temperature_{Bypass} - Temperature_{Return}\right)}$$

Note that the percentage is expressed as a decimal; for example, $1 = 100\%$ and $.8 = 80\%$

Conservation of Mass and Energy

Doing the Algebra

You can also use this concept to predict the outdoor air percentage based on outdoor air, return air and mixed air temperatures:

$$\%_{Outdoor\ Air} = \frac{1 \times (Temperature_{Mixed\ Air} - Temperature_{Return\ Air})}{(Temperature_{Outdoor\ Air} - Temperature_{Return\ Air})}$$

Note that the percentage is expressed as a decimal; for example, $1 = 100\%$ and $.8 = 80\%$ and that this assumes fully mixed air.

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Note that the percentage is expressed as a decimal; for example, $1 = 100\%$ and $.8 = 80\%$ and that this assumes fully mixed air.

Or, you can predict a mixed air temperature given outdoor and return air temperatures and percentages:

$$Temperature_{Mixed\ Air} = \frac{(Temperature_{Outdoor\ Air} \times Flow_{Outdoor\ Air}) + (Temperature_{Return\ Air} \times Flow_{Return\ Air})}{(Flow_{Mixed\ Air})}$$

Conservation of Mass and Energy
Doing the Algebra

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Or, an outdoor condition that will produce a specific mixed air temperature:

$$Temperature_{Outdoor\ Air} = \frac{(Temperature_{Mixed\ Air} \times Flow_{Mixed\ Air}) - (Temperature_{Return\ Air} \times Flow_{Return\ Air})}{(Flow_{Outdoor\ Air})}$$